On the Combustion and Heat Transmission in a Glass Tank Furnace

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The present paper was compiled from the body of data obtained in authors' laboratory. Since no single paper can hope to cover the whole important fields for the problems on the combustion and heat transfer in a glass tank furnace, the authors restricted the contents to some selected items; viz. (1) the length of gas flames, (2) the distribution of flame temperature and the evaluation of the so-called furnace temperature, and (3) the influence of convection current on the heat transfer in molten glass.

The temperature distribution in a turbulent diffusion flame formed by gas and air issuing from separate ports was investigated. A simple formula for calculating the average temperature in vertical sections at every point along the axis of flame travel was presented.

In connection with the actual furnace operation the above formula was applied for the prediction of the influences of the calorific value of fuel, and of the preheating temperatures of gas and air on the flame temperature which closely relates with the output of a tank.

By means of model experiments using gas and water as medium the flow of gases in a tank was investigated, in order to visualize the influences of the velocity ratio of gas and air as well as the circulation of combustion products under crown ceiling on flame stability.

Owing to the complexity of flow it was not possible to estimate exactly the contribution of the convection currents to the temperature distribution in molten glass. However, at least for some specified points, namely at the hot spot or others where the currents may be regarded as flowing in vertical direction, the heat transfer along these lines was studied with the results indicating that the effect of convection current should not be overlooked.

INTRODUCTION

Ever since our ancesters have learned the method of glass melting, combustion and heat transfer have been the problems of vital importance, and it will be so in the future as long as glass is produced by melting process. In recent years furnace rating is becomming increasingly important, for simple calculation will prove that the cost for producing per ton of glass would be reduced with the higher rating until at the unbelievable short term of campaign caused by the curtailed life of the refractories the gain by high pull rate would just balance with the loss due to the more frequent repair.

It is not possible, however, to increase the pull pate simply by increasing the amount of charge and pull, because the time being permitted for melting and refin-

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ing would become too short for turning out the products of the same quality. Higher rating may only be achieved through the elevation of the furnace temperature with the additional fuel supply. Recent improvement of refractories has constituted valuable contributions to prolong the terms of campaign permitting to operate the furnace safely at higher temperature, and the operating temperature of modern furnaces hikes up higher and higher.

As the heat input to a furnace from fuel and preheated air for combustion is in balance with the sum of heat being used for melting and refining the glass and the heat loss from various sources the amount of fuel which should be supplied to the furnace may be estimated assuming that this amount could be used precisely in accordance with the scheme given by heat balance.

This, however, is the mere assumption resting on no authority whatsoever, and there are many cases in which the supplied heat can not furnish the necessary amount of heat for melting and refining. The study of the heat balance of a furnace system is very important for the diagnosis of an existing furnace. It also may be used for improving the furnace design, since our past experience provides the very good guide posts so that we can make the best use of the results of our diagnosis.

The second approach to the problem is to find out the correlations between the governing factors from the results of direct measurements.

Sharp[1], Lyle[2], and Shwarbe[3] are among those who have published the results of this kind of investigations. Sharp, for example, formulated the experimental results obtained by the school of Turner to a simple equation which gives the temperature being necessary to guarantee a certain rate of pull. He also found out that the rise in melting temperature necessary for an additional ton of load is inversely proportional to the melting area of the furnace. These equations rest on sound experimental bases and may be used very conveniently as long as the nature of constants and the limitation of applicability are well informed.

The third approach to the subject is the aerodynamical studies of the combustion of fuel, the flow of gases, and the studies of heat transfer by radiation and convection. A relatively few papers on the combustion and heat transfer in a glass tank furnace have already been published. A few years ago Kanai[4] calculated the temperature distribution of a flame in a tank furnace, Kruszewski[5] has investigated the velocity distribution of combustion gases using a model tank, and Kellete[6] calculated the temperature distribution in molten glass. Quite recently Rummel[7] published the results of investigations on the flow and mixing using a gas model, and Czerny and Genzel[8] made public of their extensive work on the penetration of radiation in glass.

The present paper was compiled from the body of data obtained in authors' laboratory. Since no single paper can hope to cover the whole important fields the authors restricted the contents to some selected items, viz., (1) the length of gas and oil flames, (2) the distribution of flame temperature, and the evaluation of the so-called furnace temperature, and (3) the influence of convection current and of the glass composition on the temperature distribution in molten glass. In spite of this limitations the space does not allow to discuss the problems with concrete examples, and it is hoped that this paper may be able to afford some basic ideas to those who are not so familiar with the science of combustion and heat transfer.

I. THE LENGTH OF GAS AND OIL FLAMES

It is the custom of furnace operators that they watch and regulate the burning condition so that the flame always fades away just before the exhausting port. Shorter flame results poor overspreading, while longer one would bring refractories of uptake and regenerator to earier damage. Flame length is the most easily measurable and still very important characteristic.

There are many types of flames in existence as shown in Table 1. The more important type for glass technologists is the enclosed flames with limited air supply, since we are participating every day with the combustion belonging to this category. Using liquid and gas models a series of experiments have been carried out in order to elucidate some important flame characteristics.

Table 1. Classification of flames.

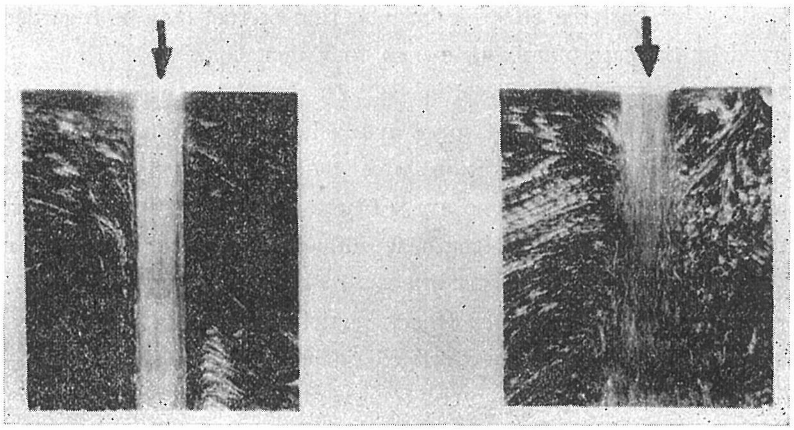
Type	Condition	Flame charcterirtics
Premixing flames	Fuel and air being mixing beforehand	The balance between the flow and the burning velocity determines the form of the combustion surface.
Diffusion flames	Combustion air is supplied by molecular or eddy diffusion	The balance between the flow and the diffusion velocity determines the form of the combustion surface.
Open flames in air (1) Molecular diffusion flame	Small Reynolds number	Flame length is proportional to the port velocity.
(2) Turbulent diffusion flame	Large Reynolds number	Flame length is proportional to port diameter being independent of the port velocity.
Enclosed flames with limited air supply	Smaller Reynolds number	Flame length is determined primarily by the excess air rate together with the size and shape of the combustion chamber.

(1) Experiments with Liquid Models

(a) The structures of open jets. Probably the experiments by a liquid model whose results are reproduced in Fig. 1 might help to make out the reason why the two different manifestations of an open flame occur under different conditions.

These are the photo-reproductions of the tracks described by the tracer particles rushing with a jet stream and moving around in the surrounding medium. Through a glass wall the jet was illuminated by a thin slab of light which cuts the stream at the center.

These figures show clearly the difference of the structure of jets having different Reynolds numbers at the nozzle. In streamline flow whose Reynolds number is as low as 840 the surrounding liquid can not penetrate into the main stream, although the viscosity drag at the boundary induces the motion of surrounding medium which,



(a) (b)

Fig. 1. Tracks of stroboscopically illuminated particles in a streamline (a) and turbulent flow (b) (after H. Jinno).
Nozzle diameter 7 mm. Size of the chamber $12 \times 12 \times 30$ cm^3
Reynolds : (a) $Re=840$, (b) $Re=3730$
Thim between interruptions : (a) 240 ms, (b) 67 ms.

however, flows parallel along the main stream. Fig. 1 (b) shows the tracks of a turbulent flow. This fugure indicates that the flow width expands with the distance from the nozzle except a small portion very near from its mouth. This is the result of the entrainment of the surrounding medium which approaches from far-way off, changing the direction, and joins to the main stream.

(b) Nature of turbulent diffusion flame. A molecular diffusion flame is characterized by the fact that the flame length is proportional to the port velocity. As long as the surrounding medium can not penetrate into the main stream mechanically the only way which produces the intermixing of fuel and air is the molecular diffusion. And, as this is independent of the port velocity its increase will permit an elementary volume of fuel in the combustion zone to cover the longer stretch before the chemical reaction completes.

A turbulent diffusion flame is characterized by the constant flame length being independent of the port velocity. In proportion to the increase of the port velocity the amount of air, or a number of air parcels, which come to join with the main stream will be increased, because the rate of momentum transfer, the potential source of the mixing, increses proportionally with the rate of fuel supply. The jostling action of fluid parcels as being indicated by the trottering tracks of the tracers in the figure above provides the increasing contact surface between the parcels of air and fuel, and as a result the amount of fuel consumption increases.

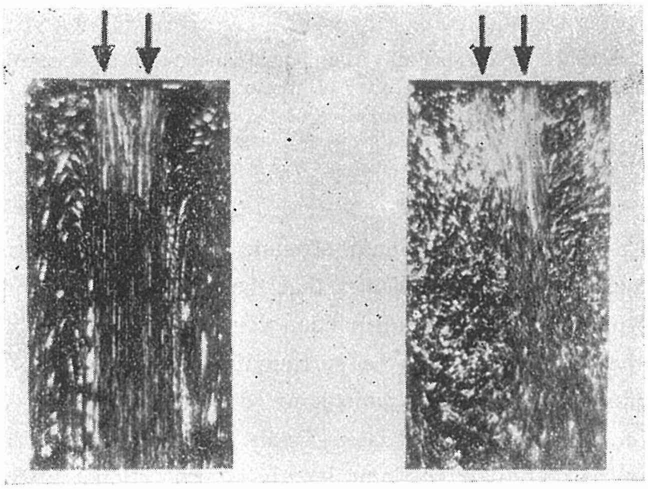
As the contact area between the fuel and air parcels increases in proportion with the increasing port velocity the flame length remains constant although its width expands. In short the effective burning rate of fuel may be regarded as if it were governed by the rate of mixing due to eddy diffusion.

(c) **Jet structure in an enclosed space.** In this case the secondary air is supplied from a separate port, and the space available for the combustion is limited.

Needless to say that these conditions affect the flame pattern. The major characteristic features of this kind of flame are, (1) the flow becomes turbulent from much lower port velocity, (2) the flame length is influenced by the excess air rate, and (3) the conditions at the exhaust port affect strongly on the flame pattern.

In Fig. 2 the tracks described by tracer particles in a narrow chamber are reproduced. The rectangular ports were arranged side by side.

When the liquid flows very slowly a stream line flow results. This state is



(a) (b)

Fig. 2. Flow patterns of a streamline and a turbulent flows in an enclosed space, (a) streamline flow, (b) turbulent flow (after H. Jinno).

Size of the enclosed space $2\times7\times20$ cm.[3] Width of ports 2 mm. Time between interruptions : (a) 170 ms, (b) 120 ms. Reynolds numbers : (a) $Re=400$, (b) $Re=800$. Port velocities : (a) same port velocity $u_g=u_a=11$ cm/sec, (b) $u_g=22$ cm/sec, $u_a=57$ cm/sec.

disturbed with the slight increase of jet velocity, i. e., when the Reynolds number exceeds a few hundred even in the laminar region for an open flame ($Re \cong 800$). Jet boundary becomes flickered due to uneven pressure distribution over the surface. From the projected points the jet liquid tends to run out while the surrounding medium tends to flow into the main stream at the depressed points. As a results the jet stream becomes turbulent from the break point to which the flow retains the pattern in the port.

As the Reynolds number of the flow in the ports of actual tank furnaces is the order of several thousands so that the flow is already definitly turbulent, and we need not worry about the position of the break point.

The ever changing circulation in the surrounding fluid, the combustion products in actual furnaces, is the source which gives the flame turning and twisting motion, and even splits it into many elementary flames which are the real burning units.

If the velocity difference exists between two jets the one having a lower velocity will be drawn into another so that after a little while both flow as a single unit as it is seen in Fig. 2 (b).

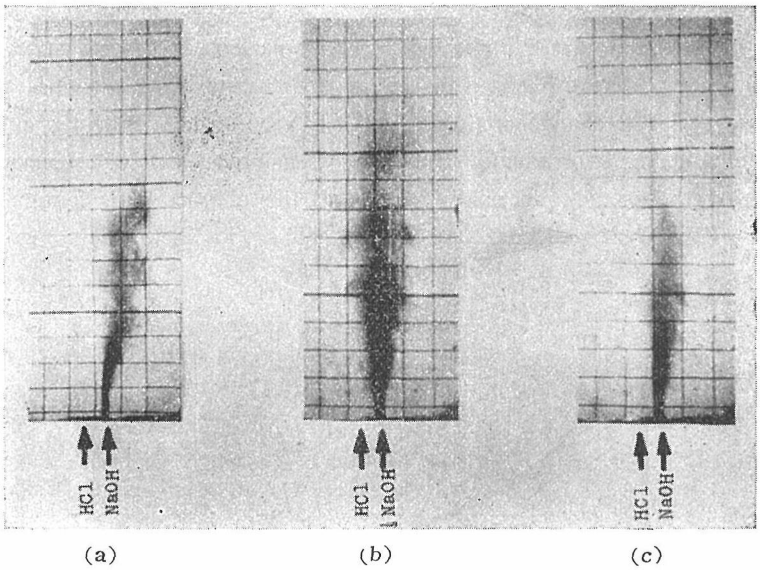


Fig. 3. Liquid model flames with limited air supply established in a confined space (after A. Ōmi).

Concentration of NaOH : (a) town gas, 0.06 N. (b), (c) Producer gas 0.024 N. Concentration of HCl : air 0.024 N.

Port velocities : (a) u_g (cm/sec)=23, (b) u_g=27, (c) u_g=27
(a) u_a (cm/sec)=43, (b) u_a=30, (c) u_a=36

Reynolds numbers : gas port (NaOH), (a) 870, (b) 970, (c) 970
air port (HCl), (a) 1630, (b) 1080, (c) 1300

Excess air rate : (a) 1.10, (b) 1.17, (c) 1.33.

(d) Liquuid model flames. So far the natures of molecular diffusion and turbulent flames established in an open air as well as in a confined space have been discussed in the light of flow patterns. However, the same problem may be approached from another angle, i. e., by a liquid model flame which permits the more direct estimation of some important flame characteristics.

In a liquid model a flame may be visualized by replacing the combustion with the chemical reaction in a liquid medium, for example, with the neutralization between acid and alkali using a proper indicator as phenolphthalein. Needless to say that the flame generated turbulence, the polymerization and dissociation of fuel etc. are not included in the model flame.

In Fig. 3 are reproduced the model flames established in a rectangular chamber of $5\times7\times30\,cm^3$, having two parallel rectangular ports at the bottom.

From the left the figures represent, respectively, town gas flame burning with the excesess air rate $\gamma=1.10$, producer gas flame burning with $\gamma=1.17$, and the same with $\gamma=1.33$. Figures (b) and (c) indicate cleary that the excess air rate is the main governing factor of the flame length.

At first sight the reason of this striking difference in length seems to be attributed to the increase of the relative velocity. This, however, is not true.

In Fig. 4 are plotted the flame length under various excess air rates. It will be seen that the points corresponding to the same excess air rates are gathering at about the same level. The change of flame length is very strong while the excess air rate is low, which decreases gradually with the increasing amount of secondary air until the flame length becomes practically constant, the characteristic feature of an open jet flame. This result indicates that the fuel air ratio should be controlled exactly

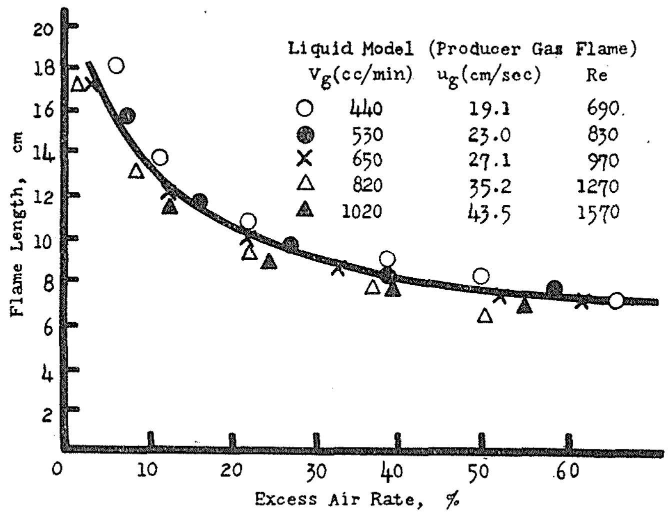
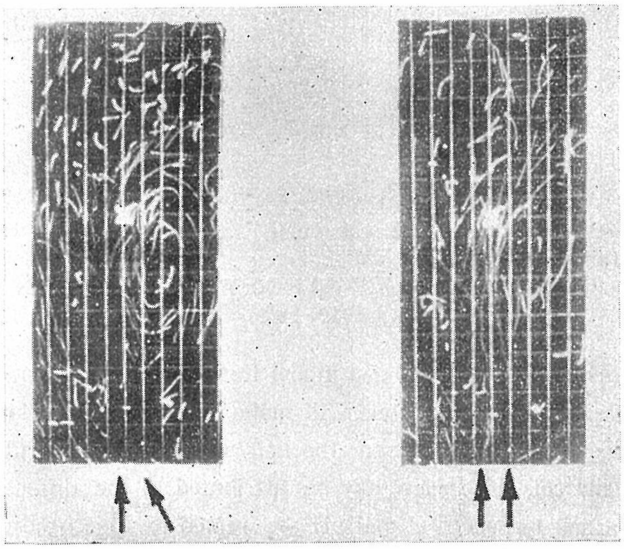


Fig. 4. Influence of the excess air rate on the flame length (after A. Ōmi). v_g : amount of NaOH, u_g : port velocity of NaOH.

in order to keep the flame length constant.

(e) **Circulations in the surrounding medium.** As already mentioned jets forced into a confined chamber induce the uneven pressure distribution throughout which is changing at every moment. The uneven pressure distribution in the surrounding medium causes the circulation. Although its velocity is comparatively low it exerts an important influences upon the pattern of burning flames in a combustion chamber.

The authors have carried out a series of experiments on the movement of liquid in a rectangular chamber when tow jets are forced into the chamber.



(b) (a)

Fig. 5. Circulations in a rectangular chamber induced by two jet streams (after A. Hirose).
Angles between jets: (a) parallel, (b) 30°
Jet velocities: (a) $u_g=u_a=100$ cm/sec
(b) $u_g=80$ cm/sec, $u_a=160$ cm/sec.

In Fig. 5 is shown the photo-reproduction of the tracks described by the tracers under two different burner arrangements, parallel and inclined. The pictures may be regarded as being selfexplanately. From the tracks it is possible to evaluate the scale and the intensity of turbulence in the surrounding medium.

(f) **Use of scale models.** The authors have pointed out that the excess air rate, and the size as well as the construction of the combustion chamber affect strongly upon the flame length. With the decreasing excess air rate increases the flame length, while it decreases, at least to some extent, with the decreasing chamber volume. As the chamber construction is not simple enough to be able to formulate in simple form the experiments with scale models seem to be the only way to know the combined result of these antagonistic influences.

Some results of the experiments with scale models have already been published

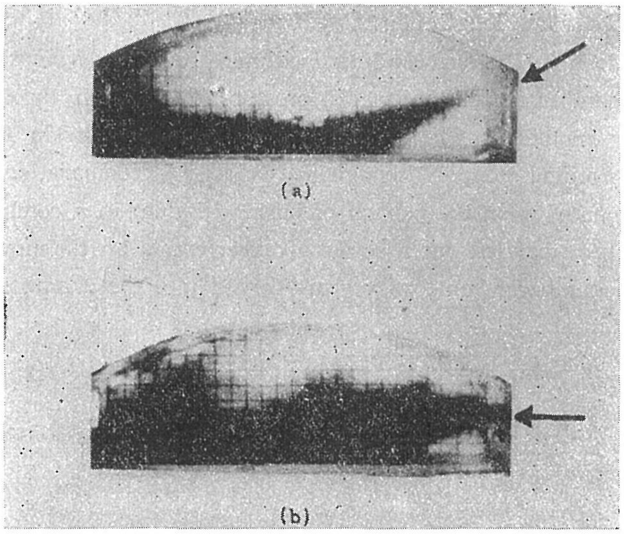


Fig. 6. Liquid model flames in a scale model, (a) oil flame, (b) gas flame (after S. Tanaka).
Port velocities (cm/sec): (a) u_g (0.04 N NaOH)=220, u_a (0.01 N HCl)=25, (b) u_g (0.01 N NaOH)=25, u_a (0.01 N HCl)=30.

by the authors[9]. Here an oil and a gas model flames are shown in Fig. 6 as representatives. Comparing (a) and (b) one will make out that an oil flame is comparatively thin, while a gas flame fills near the half way between the glass surface and the crown. Probably this difference may be attributed to the difference of the port construction. In a gas furnace fuel and air are mixed partly in the port so that the combustion begins before they enter into the combustion chamber while in an oil fired furnace atomized fuel is forced directly into the combustion chamber from a separate nozzle. As soon as the streams combine within the port the combustion permeates to all points at the periphery of the combined stream.

Also the experiments with scale models revealed that the excess air rate is the prime cause which governs the flame length, and it approaches to a constant value when the rate becomes as high as 1.5-1.6 according to the furnace and the port construction. As this is the characteristic feature of an open flame it will readily be recognized that the experiments with a scale model is a convenient mediator between the open flame and that burning in a combustion chamber. The process is as follows:

As the first step the experiments with a scale model under a certain fuel input are carried out. By increasing the excess air rate the ratio δ of the flame length of corresponding to an open jet flame to that obtained under a desired amount of air supply is worked out. For a rough estimation the temperature correction will not be necessary, because it drops off from the ratio as long as we assume that the correction term is independent of the amount of the preheated air.

The second step is to determine the length of burning flame in an open air. Concerning the open flames many excellent works have already been published. Cude,[10] for example, presented a formula which gives the length of open flames. His equation may be transformed as

$$L=\frac{kd\sqrt{\pi s'}}{2R}, \qquad R=1+r, \tag{1}$$

in order to express the length of an oil flame in terms of the nozzle diameter of the burner, d, and the ratio of primary air to fuel, r lb/lb, which allows to estimate L if the proper values of the constant k and the density of jet fluid, s', are known. Cude gave the value $k=1100\ (\mathrm{ft^3/lb})^{\frac{1}{2}}$ for an inside mixing burner, and s' may be estimated as $s'=60\ \mathrm{lb/ft^3}$.

The value $\delta\times L$ predicts an approximate length of the flame burning actually in the furnace under the given conditions.

(2) Experiments with Gas Models

We are now comming to the stage to give brief accounts of the results obtained with a gas model in which the gas is actually burning. The experiments have originally been designed in order to study the fine structure of a flame, the problems such as the intensity and the scale of turbulence in a diffusion flame. Here, however, just a few points will be touched which are closely related with flame length.

A rectangular combustion chamber, $3\times7\times30\ \mathrm{cm^3}$ was provided with a demountable burner at the bottom which has two openings for gas and air. On the side

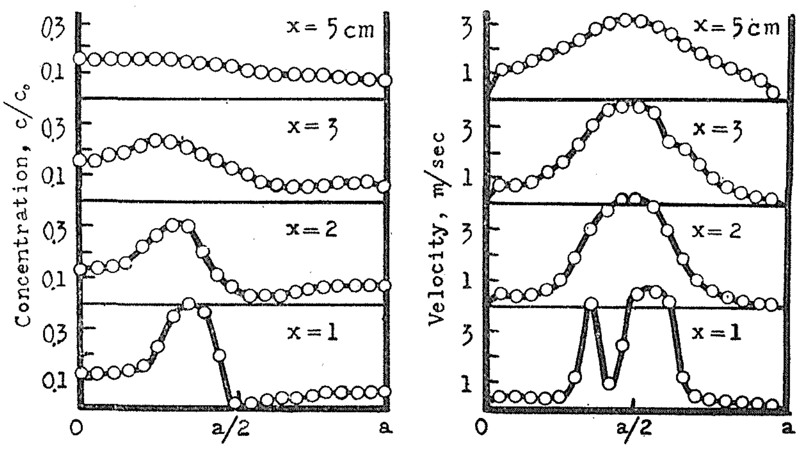


Fig. 7 (a), (b). Concentration and velocity paterns in cold gas model, (a) concentration profile, (b) velocity profile (after A. Furumiya).
Port velocity of gas $u_g=5.0$m/sec, port velocity of air $u_a=5.0$m/sec, x=distance axial from port, c/c_0=ratio of hydrogen in fluid at any point and in initial fluid, a=width of chamber.

wall are provided the sampling holes of 5 mm in diameter at 1 cm intervals. They were covered with glass plates except the one which was used for that moment.

For gas analysis the physical methods were used availing the thermal and magnetic characteristics of hydrogen and oxygen, respectively.

The apparatus is not large enough to be able to simulate the dynamical and thermal conditions of actual furnaces, because the lift and the blow off of the flame prohibit to carry out the experiments with high Reynolds numbers.

In Fig. 7 are shown the results of cold experiments representing, respectively, the concentration and the velocity patterns at different levels. In the figure, (a) represents the change of the fraction of hydrogen when the nozzle fluids flow upwards while (b) shows the velocity profiles. The concentration as well as the velocity profiles indicate clearly the leveling action due to eddy diffusion.

When the fuel is ignited (c) the concentration profiles assume the entirely different shape, because the fuel and oxygen interrupt each other to penetrate across their burning zone. At first sight, the curves, which represent the time average values,

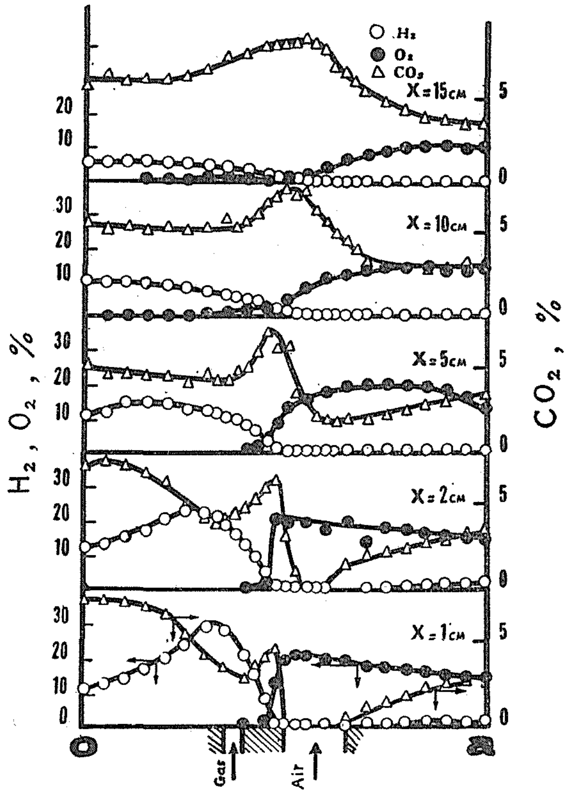


Fig. 7 (c). Concentration profile of the diffusion flame formed by city gas and air supplied separately from rectangular ports to the combustion chamber (after A. Furumixa).
Port velocity $u_g=u_a=5.0$m/sec, x=axial distance from port, a=width of chamber.

look like very much similar to those of molecular diffusion flames, which usually obtained by flowing the gas and air with the same velocity, although we are dealing with the type which was confirmed to be definitely turbulent. There are, however, the cardinal differences between these two.

By close inspection one would make out that the concentration profiles of O_2 and H_2 penetrate each other to a considerable depth, and they have maximum locating at some distances from the wall. Also the curves of CO_2 give high values especially where the jet expanse does not reach to the wall.

All these phenomena are the results caused by the turbulence occuring in the combustion chamber.

Oxygen is diluted by the combustion products both comming from the combustion zone and from the surrounding, and consequently the chance of meating the fuel parcel with that containing a sufficient amount of oxygen molecules would be reduced. In other words the apparent reaction velocity, so to speak, decreases with the vertical distance from the ports. This is the reason why the excess air rate is sensitive to the flame length.

Comparing Fig. 7 (c) and (d) it may be concluded that the combustion zone exists at the points where CO_2 assumes the maximum value.

To speak the truth we are comming only to the half way of the discussion of the flames burning in a combustion chamber. We have not touched yet at all the important roles played by the exhausting side of the furnace, which controls the furnace pressure by means of a damper. As a furnace has many openings at various points the fluctuation of the furnace pressure induces the change of the amount of

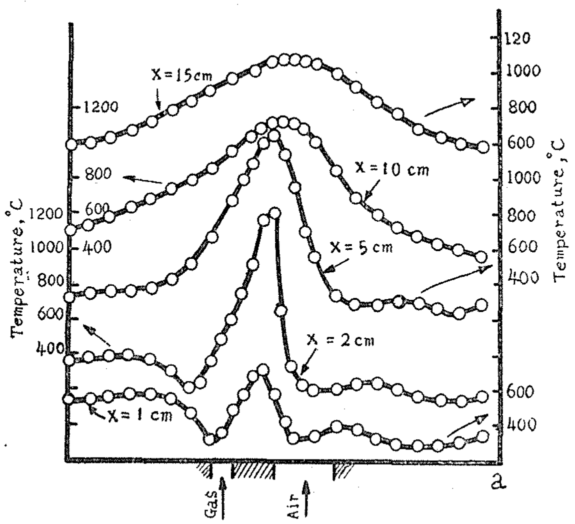


Fig. 7 (d). Temperature profiles (after A. Furumiya).
The arrows indicate the temperature axis to which the respective curve should be referred.

the infiltration from and the leakage to the surroundings, and as a result the local flow patterns change strikingly by the slight change of the furnace pressure. This is the reason why we should have to control the furnace pressure as accurately as possible.

The authors believe that this kind of problems may only be approached through the well organized gas model experiments.

Unfortunately space is not here to discuss the results of this series of experiments even in some detail. Here a figure will be presented which shows the concentration profile of H_2 in a cold experiments using a scale model (Fig. 8).

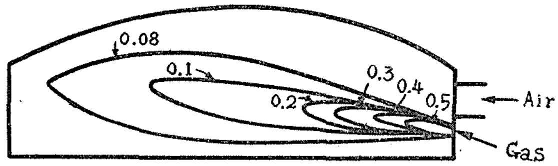


Fig. 8. Concentration distribution of hydrogen in cold gas experiment using a scale model (K. Murakami).
c/c_0=ratio of hydrogen in fluid at any point and initial fluid.

II. THE DISTRIBUTION OF FLAME TEMPERATURE

A glass tank belongs to a group of so-called bath type furnaces. Its most popular fellow member is an open hearth furnace which has been most thoroughly studied from substantially the same angle with our investigations. Although the mechanism of heat transfer is the same in both types the conditions which should be fulfilled for a glass tank are basically different from those being most important for the operation of an open hearth furnace.

In a glass tank the temperature and its distribution is of prime importance, since the former determines the allowable output, while the latter affects immediately the quality of the product. There are two main factors which have an important bearing upon the temperature distribution, one the charging of batch and the other the heat transfer from flame to glass. First of all the temperature distribution in a flame will be discussed.

(a) **Change of the concentration of combustibles in a diffusion flame.** In a diffusion flame the heat generated is discharged continuously to the environment by radiation and convection so that the flame temperature at a point is determined by the balance between the rates of heat generation in, and of discharge from an elementary volume at this point. The temperature, therefore, can not be estimated without knowing the amount of heat being discharged to the environment. There is, however, a key for solving this problem. The problem may be approached from the estimation of the concentration profile of combustibles which remain in the burning flame until the last moment when the flame fades away.

A few years ago[11] the authors advanced an equation for the purpose of calculating the two dimentional concentration profiles of a turbulent flame of burning gas in a confined space. The results came in essential agreement with those of model experiments proving that the equation, originally designed for two dimentional patterns, is also available for the evaluation of the patterns in an actual flame which justifies for rough estimations an assumption that the profile remains practically unchanged throughout the whole thickness.

For the present purpose it is not necessary to be deeply concerned about the fine structure of a turbulent flame but instead it is only necessary to set up a model which permits the estimation of the average concentration at any cross section along the flame travel. As such a model an equation having so simple form as

$$\bar{c}_x = C_\infty + a\bar{C}_0 e^{-K'x} \tag{2}$$

may be used, where $\bar{c}_x$ is the average concentration at a point x along the flame travel, C_∞ the concentration at infinity, $\bar{C}_0$ the average modified initial concentration of combustibles. The constant a varies with the port arrangements. The term K' is called as the combustion rate coefficient which is concerned in the rate of fuel consumption estimated along the line of flame travel.

The value of K' varies not only with the excess air rate and the furnace construction but also with the velocity ratio of fuel and air as well as with the air volume. Furthermore K' is the function of x.

Although the authors believe that the way which allows the exact evalution of K' is an experiment with a gas model whose results being corrected for the temperature effect there is another avenue opened for the rough estimation of an average value.

The length of a flame burning in an actual tank furnace operating under certain conditions may be predicted by putting together the results of model experiments with those for an open jet flame. Under a specific burner arrangement and fuel input the open flame burns with the combustion rate coefficient K', while it burns with another combustion rate coefficient K in the combustion chamber, and consequently we can treat an enclosed flame as if it were burning in an open air with the combustion rate coefficient K.

The value K for this hypothetical open flame may easily be obtained by substituting the flame length modified with the procedures mentioned in I (f) into the well known equation of turbulent open flames viz.,

$$\bar{c}_x = c_0 e^{-Kx}\ , \tag{3}$$

c_0 is the initial concentration of combustibles, $\bar{c}_x$ the concentration at any point x. Needless to say that an arbitrary value for $\bar{c}_x$, say $\bar{c}_x/c_0=0.05$, should be assigned to the concentration at the flame tip.

Once K is known it is easy to calculate the concentration distribution along the whole flame length.

(b) Evaluation of temperature distribution in a flame. A good many years ago the authors have posturated a formula which gives the temperature distribution in a turbulent flame[12]

$$\frac{dt_f}{dx}=\frac{q_0Kc_ve^{-Kx}-dq_v}{\sum(c_pv)}, \tag{4}$$

without having deep views of the nature of K. In the equation t_f is the mean flame temperature at any cross section along the line of flame travel x, q_0 the heat value of the fuel, c_v the fuel consumption, dq_v the heat given out to the surroundings, $\sum(c_pv)$ the heat content of the combustion products referred to the mean specific heat at the preestimated temperature.

The numerator expresses the balance between the heat evolved from a hypothetical flame having the same combustion rate coefficient K and the heat discharged to the glass surface as well as to the superstructure of the furnace.

The second term dq_v may be expressed as

$$dq_v=\sum 4.96\phi A\left[\left(\frac{T_f}{100}\right)^4-\left(\frac{T_E}{100}\right)^4\right]+\sum h(T_f-T_E), \tag{5}$$

in which T_f and T_E are, respectively, the temperature in °K of the flame and mean temperature of the surroundings, for example, of the crown ceiling and the glass temperature, h the heat transfer coefficient for convection, A the area of the surroundings, and ϕ is the effective emissivity taking into account of the geometrical factor as well as the emissivities of the surroundings.

Substituting the equation (5) into (4) and assuming the constant glass temperature throughout, and the constant temperature difference between the flame and the superstructure along x axis, t_f may be estimated by numerical integration if the property of fuel and the operating condition of fuel are known.

The evaluation of q_v is possible from another angle, namely, from the heat balance of a tank furnace as a whole system. The method will be illustrated in the next section. Once the average value of the heat loss $H_R=H_1+H_2$ referred to axis x is known the equation (4) becomes readily integrable.

Hence the flame temperature t_f at any point along the line of flame travel x will be given by

$$t_f=\frac{q_p+q_0C_v(1-e^{-Kx})-H_Rx}{\sum(C_p\cdot v)}, \tag{6}$$

where q_p is the sensible heat in the secondary air and fuel, and the other notations are the same as before. It is said that for a horseshoe flame the length L should be taken as 1.5 times of the span of the furnace. Hence it is also possible to find out the proper value of K through the processes illustrated before.

(c) **Average furnace temperature.** So vaguely is used the word "furnace temperature" among glass technologists. It is meant either as the surface temperature of molten glass or the temperature of burning flames. Such temperature, however, differs from point to point in the furnace, and this is even the characteristic feature of a glass tank which is being closely related with the furnace rating and the quality of products. Moreover, this is the point by which a glass tank is recognized all its own among its fellow members, the bath type furnaces.

Let us try to estimate the average flame temperature taking as an example a cross fired tank of 7.5×5 m^2 having three ports.

The tank is assumed to be operating with the pull rate of 40t/day which is fired by 940 Nm^3/hr. port of the producer gas having the net calorific value of 1470 Kcal/Nm^5 with the excess air of 10%. The average preheating temperature of gas and air is assumed to be about 1100'C.

With the walls, being inpermeable to heat, the furnace is supposed to be divided into three compartments parallel to the backwell at the mid points between the ports. To the glass surface confined in each compartment an appropriate mean temperature is assigned, which may easily be picked up from the existing date of direct measurements[13].

In this case such temperatures are 1420, 1470, and 1430°C respectively, putting in the order from the doghouse.

Then the average value of the inner surface of superstructure plus that of the glass surface is used as the next best substitute in order to reduce the problem to that of the heat transfer from a flame filling the two infinitely extending parallel plates. If it is necessary to consider the furnace system including the side wall of the lower

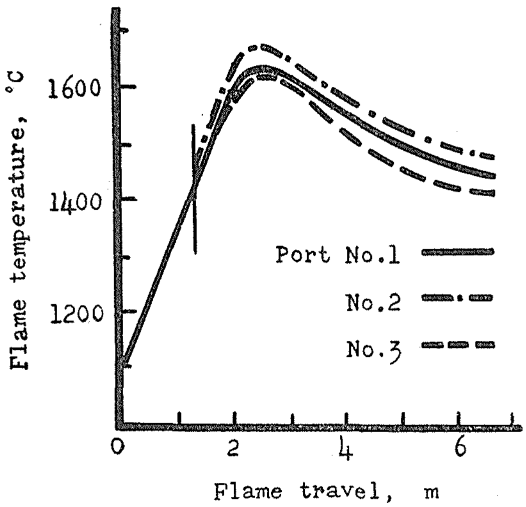


Fig. 9. Temprature distribution of flames in a glass tank furnace.

structure containing molten glass as in the case being delt with in the next section the average value the whole area concerned should be taken.

The next step is to assign the proper values in the equation (5) and to assume the temperature difference between the inner wall and glass surfaces, for example, $\Delta t=30°C$. The second term representing the heat transfer by convection was neglected in the present calculation, for it is generally known that it is of second importance.

The results obtained are reproduced in Fig. 9 which gives the average flame temperature $t_{Fm}=1540°C$. Moreover, it is seen from the figure that the flame temperature for No. 2 port is the highest whose maximum is locating somewhat nearer to the port. As this process does not include the heat balance of the whole furnace system the mean flame tempreture obtained under given amount of fuel is available for keeping the check on the assumed mean flame temperature which is necessary for the evaluation of the amount of fuel supply from the heat balance of the furnace. Furthermore, the estimation of the position of the maximum is important for the radiamatic control of the furnace temperature, because we usually measure the highest temperature which should be located on the centerline in order that the reading does not fluctuate before and after the reversing of the regenerators.

(d) Evaluation of the necessary fuel supply. Obviously, the heat input to a furnace from fuel and the preheated air are in balance with the sum of heat used for melting and refining the glass and the heat loss originating from various sources, so that the fuel consumption G may be represent by

$$G=\frac{H_1+H_2}{q_o-Q(1-E)} \quad (\mathrm{Nm^3/day}), \tag{7}$$

where q_o is the calorific value of fuel, E the efficiency of regenerators, Q the heat being carried out from the furnace by the combustion products, H_1 the heat required for melting and refining the glass, H_2 the gross heat loss from the furnace walls.

Suppose a tank whose melting area is A ($\mathrm{m^2}$) operating with the pull rate P ($\mathrm{kg/m^2}$, day) H_1 may be expressed as

$$H_1=(q+C_pt_s)A\cdot P\ , \tag{8}$$

in which the terms in the parenthesis represent the amount of heat referred to per kg of glass. The amount of heat required for glass formation q may be estimated from the data recently published by Kröger[15], C_p the mean specific heat of glass up to t_s which may be picked up from the current publications as long as the proper value of t_s is known.

A few words would be necessary for the proper interpretaton of t_s. We the glass technologists are accustomed to use the word "glass temperature" in many different ways without making any distinction. Needles to say the temperature of molten glass in a tank furnace is different from point to point. To which temerature corresponds t_s? The authors rather prefer the mean surface temeature of molten glass for t_s, since all molten glass is bound to be heated up at least once near to the surface temperature by virtue of the convection currents, which is also the necessary condition for obtaining the well refined glass. In other words t_s is the minimum temperature to which the glass should be heated up in order to obtain the well refined glass. The rough estimation of the maximum temperature required for melting the glass of given batch composition will aford an approximate value of t_s which may be modified for a tank furnace by the formula given by Lyle[2].

As the influences of some minor factors have been neglected the above presumption is not perfectly correct, although it would be nearly right for the present purpose.

In order to evaluate the second term H_2 let us substitute infinitely extending layers of flame and glass bounded by refractory walls for the actual furnace system. The wall thickness as well as that of flame and glass are assumed to be equal to the corresponding figures of the actual furnace. Let us confine our attention to the heat flow normal to the plane boundary, and for this purpose let us use the mean flame temperature t_{fm}, the mean temperature of glass surface t_s, and the mean wall temperatures. Having thus excluded the heat flow parallel to the surfaces the average area A as illustrated in the preceding section will be introduced.

As H_2 is composed of the heat losses, one from the super structure, and the other from the molten glass through the side and bottom blocks it may be expressed as

$$H_2 = A[U_c(t_{fm} - t_o) + U_g(t_{fm} - t_o)] \ , \tag{9}$$

if the over-all heat transfer coefficient U_c in the direction of the upper wall, and U_g in the direction of glass surface, and the temperature of the surroundings t_o are known.

U_c is represented by three terms, namely,

$$\frac{1}{U_c} = \frac{1}{h_c} + \frac{l}{\lambda} + \frac{1}{h_{rc}} \ , \tag{10}$$

where h_c is the coefficient of heat transfer from flame to the inner wall surface, and h_{rc} is the outside surface film coefficient of heat transfer at the outer surface of the wall, whose thickness and thermal conductivity are d and λ, respectively. λ should be modified taking into consideration of the simplifying procedure.

The value of h_{rc} will be found out in the text books if the approximate values of the temperature of the wall and the surrounding are assumed.

h_c may be estimated using the simplified formula

$$h_c = \psi \cdot 1101 \ , \tag{11}$$

which is valid for the temperature range of 1300–1600°C, and $\psi = 0.22$ taking the emissivity of the flame as 0.3.

For U_g the similar relation may be applied, namely,

$$\frac{1}{U_g}=\frac{1}{h_{rg}}+\frac{1}{h_g}+\frac{l'}{\lambda'}+\frac{1}{h'_{rg}} , \tag{12}$$

where h_{rg} is the coefficient of heat transfer from flame to glass surface, h_g of the heat transfer through molten glass by radiation, conduction, and convection, and h'_{rg} is the outside surface-film coefficient of the outer surface of the bottom wall having the thickness and the thermal conductivity l' and λ'.

All terms other than h_g in the right hand side of the equation may be estimated in the same way as before.

The evaluation of h_g, however, is the another case. The contributions of radiation and conduction may be put together into a single term h_{g1} in the same fashion as Kellett so that

$$h_{g1}\cong\frac{\beta}{D}b \quad , \qquad (b\cong0.79) , \tag{13}$$

where D corresponds to the depth of the tank, and β is the radiation conductivity after Kellett and b is a correction factor being introduced by the use of the average area A. Although β varies in proportion to the third power of the glass temperature in °K it will be nearly right to use the value comming under the surface temperature t_s, since the temperature decreases rather slowly except the thin surface layer. Furthermore, β is strongly affected by the chemical composition, especially by the amount of FeO included in a colourless glass as it has been published by one of the authors[16]. Needless to say some colouring agents affects vitally. In fact β for a colourless glass was found to be 66, while for a green glass it assumes the value being as low as 10 (Kcal/mhr°C), both at 1450°C.

Having determined the contribution of radiation and conduction, h_{g1}, we are now comming to the final stage, the evaluation of the contribution of convection currents, h_{g2}. Roughly speaking the major part of h_{g2} represents the increments of the heat discharge from the side wall of the tank due to the transportation of hotter glass mass to the layer next to inner wall surface as well as to the radiation and conduction in the direction parallel to the glass surface. An additional amount of fuel should be supplied in order to cover this convection loss. An equation has already been derived which gives the heat flow under the coexistence of the mass flow of glass when the direction of flow is parallel to the side wall. In order to use this relation a hypothetical circulation repesenting the total effect of the convection current was set up, and considering a cycle composed of the two heat receiving, and two heat discharging steps h_{g2} was worked out. Owing to the lack of space only the final result will be presented, viz.

$$h_{g2}\cong k'a\beta A'/A \quad , \qquad k'\cong\frac{\rho C_p u_s}{\beta} , \tag{14}$$

in which A' is the area of the glass surface, and k' is a constant being proportional

Table 2. Terms for estimating the heat loss from a tank furnace.

	Nomenclature*	Unit	Numerical value	
Loss from super-structure	A	m^2	63	
	U_c	$Kcal/m^2hr^\circ C$	5.94	
	t_{fm}	$^\circ C$	1540	
	t_o	$^\circ C$	30	
	$AU_c(t_{fm}-t_o)$	Kcal/hr	5.65×10^5	
Loss through molten glass			Colourless glass	Green glass
	h_{rg}	$Kcal/m^2hr^\circ C$	242	242
	λ'	$Kcal/mhr^\circ C$	2.3	2.3
	ι'	m	0.3	0.3
	h'_{rg}	$Kcal/m^2hr^\circ C$	19.0	19.0
	β	$Kcal/mhr^\circ C$	66.0	10.0
	b		0.79	0.79
	α		0.34	0.34
	k'	m^{-1}	1.7	1.7
	h_g	$Kcal/m^2hr^\circ C$	76.0	11.5
	U_g	$Kcal/m^2hr^\circ C$	4.98	3.70
	$AU_g(t_{fm}-t_0)$	Kcal/hr	4.74×10^5	3.50×10^5
	H^2	Kcal/hr	10.4×10^5	9.15×10^5
	Hc**	Kcal/hr	1.45×10^5	1.07×10^5

* The meaning of above nomenclatures is referred to the appendix.
** Amount of heat used for convection currents.

to the product of the density ρ, the specific heat C_p, and the average flow velocity u_s of glass at the hot spring, and α is a constant introduced in the process of the calculation. If the temperature at the spring is assumed C_p and ρ are easily picked out from current publications, and u_s may be evaluated from the hypothetical cycle. Using the data given by Trier[13] k' and α have been worked out with the result $k'=1.7m^{-1}$, and $\alpha=0.34$, assuming the thermal conductivity of the refractory λ' is equal to 2.3 Kcal/mhr°C and the result of the estimation of u_s are 0.03, 0.17 m/hr for green and colourless glass respectively.

It was disclosed that the heat loss through molten glass is composed of two factors, one the loss by radiation and conduction which is closely related with the glass composition, and the other by convection currents which is governed by the properties of glass at the hot spring and the mean velocity u_s at this point. To speak the truth the last term does not belong to the loss but to the useful heat which should be substructed from the total heat loss.

The heat balance thus computed is summarized in Table 2.

To H_2 a few per sent of the total loss should be added as an unaccounted heat loss.

Thus H_1 and H_2 are known. And, as Q in the equation (6) may be evaluated from the known fuel composition and the excess air rate the necessary amount of fuel supply G may be calculated by assuming the proper value for E.

SUMMARY AND CONCLUSION

To summarize the above statements let us illustrate how the above equations are used for solving some practical problems.

Let us imagine that we have a furnace design at hand or a furnace is operating under certain conditions, and suppose that we are going to diagnose the furnace.

1. Pull rate P is determined by the required quality of the product.
2. Mean glass temperature t_s is known from the result of pot tests.
3. Mean flame temperature t_{fm} is assumed.
4. Probable total heat loss is estimated considering separately the heat loss from the super structure and the loss through the tank blocks.
5. Necessary amount of fuel supply is calculated.
6. Proper value of the combustion rate coeffient K is worked out from the results of model experiments and the length L of an open flame.
7. Knowing q_v and H_R the temperature distribution in the flame is calculated, from which mean value is figured out in order to compare with the assumed value of t_{fm}.
8. The mean temperature distribution thus obtained is compared with that worked out by assuming a certain glass temperature for each burner, because the mean temperature distribution obtained by both methods should be in reasonable agreement.

To speak the truth the above calculations are the sequence of trial and error being laborious if not difficult. By going through this avenue, however, we can collect some valuable informations which will be useful for establishing a sound basis for the design and the operation of furnaces.

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NOMENCLATURE

a : constant in Equation (2)
A : average area of heat-transfer surface
A' : melting area
b : correction factor in Equation (13)
c_0 : initial concentration of combustibles
c_v : fuel consumption
$\bar{c}_x$: average concentration of combustibles at a point x
$\bar{c}_0$: average modified initial concentration of combustibles
C_∞ : concentration at $x=\infty$
C_p : mean specific heat of glass
$\Sigma(C_p \cdot v)$: heat content of flue gases
d : diameter of burner
D : depth of molten glass in tank
E : heat efficiency of regenerator
G : amount of the required fuel supply
h : heat transfer coefficient for convection
h_c, h_{rg} : coefficients of heat transfer from flame to inner wall surfaces and to molten glass respectively
h_{rc}, h'_{rg} : outside susface film coefficients at outer surface of walls and of tank brockes respectively
h_g : coefficient of heat transfer through molten glass
h_{g1} : coefficient of heat transfer represented by Equation (13)
h_{g2} : coefficient of heat transfer represented by Equation (14)
H_R : (H_1+H_2) per unit length of flame travel
H_1 : heat required for melting and refining the glass
H_2 : gross heat loss from the furnace walls
H_c : amount of heat used for convection currents of molten glass
K, K' : combustion rate coefficient
k : constant in Equation (1)
k' : constant in Equation (14)
l : thickness of refractories in supper-structure
l' : thickness of tank block
L : length of flame
P : pull rate
q : heat required for glass formation

q_o : net heat value (per unit) of fuel
q_p : sensible heat in secondary air and fuel
q_v : heat given out from flame to surroundings
Q : heat (leaving) being carried out through exit ports per unit of gas burned
r : ratio of primary air to fuel.
S' : density of jet fluid
t_f : flame temperature (°C), T_f (°K)
t_{fm} : mean flame temperature
t_s : mean surface temperature of molten glass
t_o : room temperature
T_E : mean temperature of surroundings
u_s : average flow velocity of glass at hot spring
u_a, u_g : port velocities of gas (or NaOH) and air (or HCl) respectively
U_c, U_g : over-all heat transfer coefficients in direction of upper wall and glass surface respectively
V_g : amount of gass (or NaOH)
α : constant in Equation (14)
β : radiation conductivity after Kellett
γ : excess air rate
δ : coefficient
λ, λ' : thermal conductivities of wall and tank block respectively
ϕ : effective emissivity